



INTERNATIONAL APPLICATION PUBLISHED UNDER THE PATENT COOPERATION TREATY (PCT)

(51) International Patent Classification ⁵ :

F16J 15/34

A1

(i) International Publication Number:

WO 92/07207

(43) International Publication Date:

30 April 1992 (30.04.92)

(21) International Application Number: PCT/US91/04987

(22) International Filing Date: 18 July 1991 (18.07.91)

(30) Priority data:
595,420 11 October 1990 (11.10.90) US

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(81) Designated States: AT (European patent), AU, BB, BE (European patent), BF (OAPI patent), BG, BJ (OAPI patent), BR, CA, CF (OAPI patent), CG (OAPI patent), CH (European patent), CI (OAPI patent), CM (OAPI patent), DE (European patent), DK (European patent), ES (European patent), FI, FR (European patent), GA (OAPI patent), GB (European patent), GN (OAPI patent), GR (European patent), HU, IT (European patent), JP, KP, KR, LK, LU (European patent), MC, MG, ML (OAPI patent), MR (OAPI patent), MW, NL (European patent), NO, PL, RO, SD, SE (European patent), SN (OAPI patent), SU*, TD (OAPI patent), TG (OAPI patent).

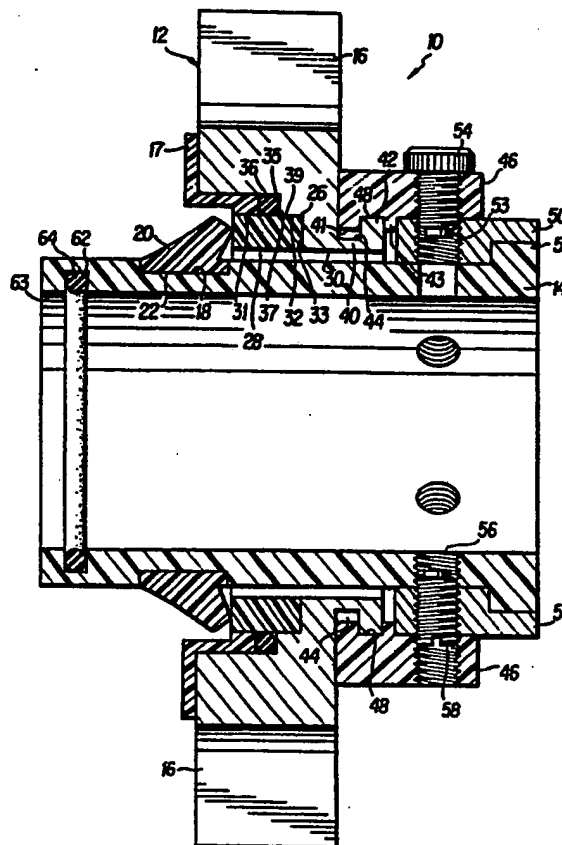
Published

With international search report.

(54) Title: ROTARY SHAFT SEALING DEVICE

(57) Abstract

A dry face seal (10) for preventing leakage around a rotary shaft (60) extending through a structural casing comprises an annular stationary rigid seal (26) and a rotary seal (20). The stationary seal, secured to the structural casing for surrounding the rotary shaft, has a smooth sealing surface for establishing a dry face seal. The resilient rotary seal means is adapted to be secured to the rotary shaft and has an annular projecting edge means (75) for containing the smooth sealing surface under a biased pressure to form a dry face seal in conjunction therewith. The resilient rotary seal means comprises a carbon filled thermally resistant polymer composition having a durometer hardness of from about 85 to about 91, and preferably comprises a fluorocarbon filled with finely divided carbon black.



+ DESIGNATIONS OF "SU"

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TITLE OF THE INVENTION
ROTARY SHAFT SEALING DEVICE

RELATIONSHIP TO COPENDING APPLICATION

5 This application is a continuation-in-part of copending application Serial No. 07/595,420 filed October 11, 1990.

FIELD OF THE INVENTION

10 This invention relates to sealing devices for rotating shafts and more particularly to seals located between a rotating shaft and a housing of a pump, pressure vessel or the like in which fluid is contained under pressure. Such fluids may includes liquids, gases, or slurries such as those containing corrosive chemicals.

BACKGROUND OF THE INVENTION

15 In industries with manufacturing processes involving fluids, numerous pumps and other rotary shaft devices are required for the transport and handling of fluids such as slurries and chemical solutions. Prior to this invention, such pumps had liquid lubricated rotary shaft seals requiring a continuous flow of water as a lubricant and sealing liquid across the seal face. This large volume of water represents a major manufacturing problem, incre. ng
20 the total water required for the process and the volume of liquid requiring treatment to make it environmentally safe be a being discharged into the environment.

25 Conventional prior art rotary seals use two opposed . faces with flat + es, one face moving across the other. The opposed surfaces are pressed together under pressure to minimize the space available for leakage. To reduce friction between the faces, water is introduced so that it flows across the face. To prevent invasion of process liquid into the bearings, water was usually introduced under pressure either into the gland seal or the stuffing box
30 lantern ring port, the pressure being greater than the pressure on the process fluid side of the seal. Water is thus directed to flow across the seal face into the process liquid, preventing flow in the

-2-

reverse direction across the seal faces and into the bearings or other parts of the device.

5 The flushing action required a large, continuous water supply. The usual amount of water introduced into a single pump stuffing box was about 2 to 5 GPM, that is, from 1,000,000 to 2,500,000 GPY.

Secondly, in most industrial plants, through operational errors or equipment failure, the supply of flushing water to the seals was interrupted, causing rapid seal failure.

10 Using such prior art seal devices, industries such as the pulp and paper plants required large volumes of water for their operations, often an average of 30,000,000 GPD. Flush water usage for packed stuffing boxes and mechanical seals usually accounted for approximately 30% of the water used in a pulp mill and about 15% of the water required in a paper mill. The same situation occurs in
15 the mining industry.

Because the process liquids, diluted by the inflow of flushing water, represent an environmental hazard, all used process liquid must be treated before being released into the environment. Tertiary level treatment is usually required. The flushing liquid
20 volume proportionally increases the volume and cost of the process liquid requiring treatment.

In the chemical industry, major problems are also presented by the conventional seal technology. Many processes require the reaction of crystalline solids with other chemicals, producing an
25 abrasive slurry. Pumping abrasive slurries is a major challenge. The chemical industry has been forced to use expensive double seals manufactured from exotic metals. The double seals are required to satisfy two requirements. They maintain pressure over the system, using a suitable barrier fluid to lubricate the faces of
30 conventional rotary seals from the inside to the outside. They reduce the radical emission produced by the rotating faces of the seals, and prevent entry of atmospheric gases into reaction mixtures.

-3-

5 New E.P.A. requirements limit radical emissions, creating a serious obstacle to continued use of conventional seals, because they require lubrication of the seals with the chemical product. Heat generated by the seals causes evaporation of products and solvents and escape (radical emission) of gases to the atmospheric side of the seal.

10 The chemical industry has attempted to solve this problem using double seals. However, the E.P.A. now requires that unless a barrier fluid is contained in a closed loop system and disposed of in accordance with contaminated liquid treatment regulations, the use of double seals will not be permitted.

The object of the present invention is to provide a solution to the above problems, particularly for industries using pumps and other devices for pumping and handling slurries and chemicals.

15

SUMMARY OF THE INVENTION

20 A unseated dry face seal for preventing leakage around a rotary shaft extending through a structural casing comprising an annular stationary rigid seal and a rotary seal. The stationary seal, secured to the structural casing for surrounding the rotary shaft, has a smooth sealing surface means for establishing a dry face seal. The resilient rotary seal means is adapted to be secured to the rotary shaft and has an annular projecting edge means for contacting the smooth sealing surface under a biased pressure to form a dry face seal in conjunction therewith. The resilient rotary seal means comprises a carbon filled thermally resistant polymer composition having a durometer hardness of from about 85 to about 91 and preferably not above 90. The thermally resistant polymer composition preferably comprises a fluorocarbon filled with finely divided carbon black.

30

Preferably, the annular contacting edge means of the resilient rotary seal means is a flat surface having a surface area greater than the contact area of the annular contacting edge means before seating occurs. The resilient annular ring has a inner base portion

-4-

adapted to be securely held in an annular mounting groove. The ring has a pivot point on a distal surface at the outer edge of the base portion. The proximal surface of the annular ring comprises outer and inner concentric proximal surfaces which converge to form an annular proximal sealing edge, the inner proximal surface extending from the sealing edge to a torque point on the proximal distal surface of the base portion.

Preferably, the angle formed by a line from the pivot point and the torque point in a plane through the central axis of the seal is within the range of about 16 to 22°, optimally from 17 to 21°. The outer concentric proximal surface forms an angle of from about 40 to 50° with a line parallel to the central axis of the seal in a plane through said central axis, and the inner concentric proximal surface forms an angle of from about 55 to 73° with a line parallel to the central axis of the seal in a plane through said central axis.

Optimally, the rotary seal element has a radially outward, forwardly diverging distal surface extending from the pivot point, and the shortest distance T, in millimeters, between the torque point and the distal surface in a plane through the central axis of the seal is provided by the formula:

$$T = \alpha + \beta ID$$

wherein α is from 3.2 to 3.6,

β is from 0.040 to 0.052, and

ID is the inner diameter of the proximal sealing edge, in millimeters.

Optimally, the rotary seal element distance D, in millimeters, of a straight line normal to the central axis of the seal from the proximal sealing edge to the surface of an axially concentric cylindrical surface through the torque point is provided by the formula:

$$D = \gamma + \delta ID$$

wherein γ is from 1.7 to 2.5,

δ is from 0.014 to 0.021, and

-5-

ID is the inner diameter of the proximal sealing edge, in millimeters.

BRIEF DESCRIPTION OF THE DRAWINGS

5 Fig. 1 is a prospective view of a sealing device embodying the principles of the present invention, shown assembled and ready for installation.

Fig. 2 is a view in elevation and in section taken along the line 2-2 in Fig. 1.

10 Fig. 3 is a view in elevation and in section showing the sealing device of Fig. 2 as it appears installed and fully seated in a typical operating installation.

Fig. 4 is a bottom plan view of a preloading clip for the sealing device of Fig. 1 and Fig 2.

15 Fig. 5 is a fragmentary cross-section of a rotary seal of this invention.

Fig. 6 is a fragmentary cross-section of the rotary seal of Fig. 5 in a relaxed condition in an initial contact with the opposing stationary seal surface.

20 Fig. 7 is a fragmentary cross-section of the rotary seal of Fig. 6 in the fully biased condition before installation on a drive shaft of a rotary device.

Fig. 8 is a fragmentary cross-section of the rotary seal of Fig. 8 after formation of the rotary sealing surface seated against the opposing stationary seal surface.

25 Fig. 9 is a fragmentary view in section showing elements of the sealing device of Fig. 2 before the rotary sealing element is preloaded.

30 Fig. 10 is a fragmentary view in section showing elements of the sealing device of Fig. 9 after the rotary seal element is preloaded and held by a clip member.

Fig. 11 is a fragmentary view in elevation and in section showing a modified form of sealing device embodying principles of the present invention.

Fig.12 is a fragmentary view in elevation and in section showing another modified form of sealing device embodying principles of the present invention.

5 Fig.13 is a view in elevation and in section of another modified form of sealing device according to this invention.

DETAILED DESCRIPTION OF THE INVENTION

10 The devices of this invention are suitable for use with rotary shaft devices used in process and mining industries. For purposes of example and not by way of limitation, the invention is described hereinafter using an embodiment which is particularly adapted for use with slurry pumps. The same basic configurations can be used for other pumps used in chemical manufacturing, and other industries requiring pumping of suspensions and chemical process solutions.

15 Referring to Figs. 1-3 of the drawings, Fig. 1 is a prospective view of a sealing device embodying the principles of the present invention, shown assembled and ready for installation. Fig. 2 is a view in elevation and in section taken along the line 2-2 in Fig. 1. Fig. 3 is a view in elevation and in section showing the sealing device of Fig. 2 as it appears installed and seated in a
20 typical operating installation.

In general, the rotary seal device 10 comprises an annular plate-like housing or gland 12 through which extends a sleeve 14 that is adapted to be mounted, fixed to a pump shaft which is rotatable about its axis. The annular housing 12 is attached to a
25 wall in a pump housing, stuffing box or a similar structure by means of bolts which may extend through holes or slots 16 in an area near the periphery of the housing. A gasket 17 is provided to seal the plate-like housing 12 against leakage.

30 Sleeve 14 can be made of any material which provides the requisite physical properties and chemical resistance, has a high operating temperature rating and is easily machined to close tolerance. An example of a suitable material having these properties is polytetrafluoroethylene (PTFE).

-7-

Referring to Fig. 2 and 3, the sleeve 14 has a groove 18 which accommodates and retains an annular rotary seal member 20 ("rotary seal") having a base 22 which seats in the groove 18.

5 The rotary seal has a proximal edge 75 which abuts an annular stationary ring seal member or seat 26. This stationary ring seal 26 is mounted and retained in an annular receptor defined the cylindrical surface 32 and annular backing surface 33 of the housing 12.

10 The stationary seal 26 has a rectangular cross-section. It has an inner cylindrical surface 28 having substantially the same radius as the adjacent cylindrical surface 30 of the housing 12. It has an outer surface which is dimensioned to abut the inner flange surfaces 31 and housing surfaces 32 and 33. O-ring 36 is positioned in an annular recess defined by the flange 17 and annular surfaces 15 35 and 37 of the housing 12. The O-ring 36 is pressed against the outer surface 39 of the stationary ring seal and housing surfaces 35 and 37, preventing passage of liquid between the ring seal 26 and the housing 12.

20 The stationary seal 26 is made of a corrosion resistant, hard material which will maintain a smooth surface in continued use. A preferred stationary seal material is silicon carbide.

25 An annular end portion 40 of the housing 12 has an outer radial flange 42 with an inner diameter which is the same as that for the stationary seal 26. The outer flange 42 and opposing wall of housing gland 12 defines a groove 44 in the housing gland for receiving one end of a series of clip spacers 46. The clip spacers are preferably made of a hard relatively rigid plastic material such as nylon. As described hereinafter, the spacers 46 are used to maintain a pre-load on the rotary seal 20. They are located at four 30 or six circumferentially spaced apart locations on the sealing device 10 as shown in Fig. 1. Each clip spacer 46, as shown in Fig. 4, has a transverse groove 48 for receiving the outer radial flange 42 and a pair of opposed ridge members 41 and 43 defining the groove. The spacers 46 are attached to and extend over a sleeve

-8-

collar 50 (Fig. 10). The sleeve collar 50 is preferably made of a metal material such as stainless steel and fits around an end flange 52 of the sleeve 14. A series of threaded holes 53 are provided at 90° spaced apart locations on the sleeve collar and each pre-load spacer clip 46 is retained thereto by a cap screw 54. Between the cap screws are a second series of threaded holes 56 which extend radially inwardly to the outer surface of the sleeve 14. Set screws 58 within the holes 56 are advanced after the sealing device 10 is installed to secure the sleeve to a pump shaft 60 as shown in Fig. 3.

In the distal end of the internal wall of sleeve 14, groove 62 is provided for retaining an O-ring 64. This O-ring provides a fluid seal between the sleeve 14 and the shaft 60 when inserted therethrough. The groove 62 is spaced back from the inner end of the sleeve 14 by at least 0.10 in. so that the hydraulic force of the liquid within the casing in a typical installation will not spread the front edge of the sleeve 63 and allow leakage along the pump shaft.

In Fig. 3, the sealing device 10 is shown in its operating and fully seated condition, installed around the pump shaft 60 and fixed by machine bolts 66 to a housing or stuffing box 568 for the pump (not shown). In the typical installation shown, the fluid or slurry present within the space 70 around the shaft and the sealing device 10 within the stuffing box is under pressure. The annular gasket 17 prevents leakage at the location indicated by arrow A, the O-ring 64 prevents leakage between the sleeve 14 and shaft 60 as indicated by arrow B. Since there is no relative movement between the adjacent structural elements at position A and position B during operation of the pump, the O-rings 36 and 64 and flange 17 are effective to prevent fluid leakage from the pump chamber, even at high fluid pressure.

During pump operation, the rotary seal 20 contacts and rotates relative to the stationary seal 26, the space therebetween providing a third potential leakage location indicated by the arrow C.

The rotary seal 20 is an important element of the sealing device 10 because it provides for effective sealing between it and the stationary seal 26 for long periods of relative movement between these two elements.

5 Fig. 5 is a fragmentary cross-sectional view of the rotary sealing element 26. Fig. 6, 7 and 8 are fragmentary cross-sectional views of the rotary and stationary sealing elements at successive unbiased, biased and fully seated stages. The physical properties, configuration and angular dimensions of the rotary seal are
10 important aspects of this invention, cooperating to provide optimum operation and durability of the seals. In Fig. 5, the annular rotary seal 20 has a back surface 71 exposed to a pump chamber. Surface 171 extends from pivot corner 170 contacting the sleeve surface 172 to the outer seal edge 173. The seal's forward surface
15 comprises outer surface 174, forming an Angle E with a line parallel to the central axis of the rotary seal and sloping from the outer edge 173 to the forward edge 175. The rotary seal's forward surface of the rotary seal also includes an inner surface 176 which extends backward from the forward protruding edge 175 to the torque corner
20 177 of the cylindrical surface 178. The cylindrical surface 178 is flush with the inner surface 172 of the sleeve. The surface 176 forms an Angle F with the surface 178 in a plane normal to the central axis of the seal. A line from the protruding edge 175 to the opposite footing corner 170 of the seal forms an Angle G with a
25 line parallel to the central axis of the seal.

In general, the dimensions of the seal should be increased proportionally with the increase in size of its diameter.

30 The thickness T, in millimeters, is the length of a straight line normal to surface 171 through torque corner 177 in a plane through the central axis of the seal, measured from surface 171 to torque corner 177. The optimum thickness T can be determined by the following formula wherein ID is the inner diameter of annular projection 175 in millimeters:

-10-

$$T = \alpha + \beta ID$$

wherein α is from 3.2 to 3.6 and
 β is from 0.040 to 0.052.

The distance D is the length, in millimeters, of a straight
5 line normal to the central axis of the seal from protruding edge 175
to the surface of axially concentric cylindrical extension of the
surface 178 through torque point 177 in a plane through the seal
central axis. The optimum distance D can be determined by the
following formula wherein ID is as defined above:

$$D = \gamma + \delta ID$$

10 wherein γ is from 1.7 to 2.5 and
 δ is from 0.014 to 0.021.

In the preferred embodiment of the rotary seal 22, Angle E is
from about 40 to 50°, Angle F is from about 55 to 73°, and Angle G
15 is from about 16 to 22° and preferably from about 17 to 21°.

The rotary seal 22 is made of an elastomeric polymer having
the chemical and thermal resistance required for the environment to
which it is exposed in use. It is loaded or filled with a
sufficient quantity of a finely divided carbon filler such as Austin
20 Black to provide a durometer hardness of from 85 to 91 and
preferably about 90. One suitable elastomeric polymer is AFLAS
(Asahi Glass Co, Inc.), a peroxide cured copolymer of
tetrafluoroethylene and propylene. Other suitable elastomeric
polymers include FKM-Tetrapolymer and base resistant FKM polymers
25 (Macrotech/CDI, Humble, TX). The minimum durometer hardness of 85
is suitable for low speed rotary seals (around 800 rpm). For higher
rotor speeds such as 3600 rpm, a high durometer hardness is
necessary and about 90 is optimum. A hardness exceeding 91 may be
too rigid and brittle to provide an effective biasing pressure and
30 sufficient movement for seating the seal in many applications.

The carbon filler has several functions. It increases the
durometer hardness and resulting stiffness of the rotary seal
required for the optimum biasing pressure and movement of the seal.
It also provides at least a part of the lubricating function as

-11-

particles, positioned between opposing seal faces to reduce the contacting surface areas of the seal and thereby reducing the frictional forces therebetween. It also influences the pyrolysis products formed at the rotary seal edge 75 as it becomes flattened in use, seating against the opposing stationary seal surface as rotary seal surface 24 to form a durable dry rotary seal.

Fig. 6 shows the seal 22 in its original conformation in contact with the stationary seal 26 before application of a biasing pressure. Fig. 7 shows the distortion of the seal from the unbiased (dotted line) position to the solid line biased position by advancement of the sealing ring 26 against the rotary seal edge 75. The biasing movement can also be the opposite movement, advancement of the rotary seal axially against the opposing stationary seal surface. This biasing movement applies a pressure against the rotary seal, distorting the resilient sealing edge, and providing the biasing pressure required to seat the seals.

Fig. 8 shows the movement of the rotary seal from the dotted line fully biased position to the solid line fully seated position. During initial operation, the high pressure per surface area and resulting high friction on the edge 75 of the rotary seal 22 as it slides across the face of the stationary seal generates heat at the circular line of contact, softening and pyrolyzing the contacting surface of the rotary seal. The rotary seal tries to return to its relaxed, unbiased conformation and this movement, in combination with pyrolytic action at the contact surface of the seal, increases the contact surface to an optimum value (width H) and further hardens or cures this enlarged surface to form an effective durable, low friction dry face seal. The width H determines the effective contact surface of the rotary and stationary seals and the heat generated during continuous operation of the seal. If this area is too small, the biasing pressure is concentrated on a small area, generating excessive heat and distortion of the sealing face 24. If the sealing face 24 is too large, the biasing pressure becomes too widely distributed to maintain an effective seal without the

-12-

application of excessive pressure. The ultimate sealing surface area of the seated rotary seal is thus determined by a combination of the durometer hardness (and inherent resiliency) of the seal material and the shape and relative dimension (surface angles) of the rotary seal. As the edge 75 is transformed to the flat surface 24, the pressure per unit area decreases until the heat is dissipated at a faster rate than it is produced, allowing cooling of the contact area, cessation of pyrolysis and further distortion of the seal, and final cooling to form a hard durable low friction surface. The durometer hardness and dimensions of the rotary seal 22 are thus very important for establishing an effective durable seal.

xx Sufficient pre-load or bias pressure is retained to insure maintenance of the dry face seal between the moving seal elements. This bias pressure is reflected by a continued curve in the surface 71. This curve is important to smoothly deflect liquid from an axial to a radial direction with low friction, avoiding direct perpendicular impact of liquid and suspension components on the seal surface, thereby reducing reduce erosion of the surface. The selected rotary seal dimensions thus also control the residual curve of surface 71.

Referring to Fig. 5, the rotary seal is molded to provide a base portion 22. To secure the rotary seal 20 in a stationary position in groove 18, the radius of the molded, relaxed rotary seal 20 is selected to be less than the radius of the base of the groove 18, and the width of the base 22 in the axial direction is greater than the axial dimension of the groove 18. In position, the rotary seal 20 remains stretched and under tension, forcing the rotary seal inner surface securely against the surface of the groove 18. The radius of the rotary seal is selected to provide sufficient tension for this purpose. To compensate for the narrowing of the rotary seal in the axial direction due to stretching, the rotary seal 20 is molded so that its base portion 22, in its mounted state under tension, is sufficiently wide to slightly exceed the axial dimension

of sleeve groove 18. The width of the base 22 is molded to have the dimensions providing that when the rotary is stretched for mounting, the base 22 of the rotary will slip into the groove 18, and when the base 22 is nested fully in the groove 18, it will have expanded in the axial direction to press against the opposed faces 179 and 180 of the groove 18. Molding the rotary seal so that its base portion 22, in its relaxed state, is approximately 10 percent wider than the axial dimension of the groove 18 is usually sufficient. If necessary, a small amount of adhesive such as rubber cement can be applied to the groove 18 before mounting the rotary seal 20 to insure the rotary functions or rotates as if it were an integral part of the sleeve 14.

As shown in Fig. 9, when the sleeve 14 having a rotary seal 20 fixed in its groove 18 is first placed within an annular plate-like housing 12, the end flange 42 of the gland housing 12 is closely adjacent to the inner edge of the metal end ring 50.

As shown in Fig. 10, to "pre-load" the rotary seal 20, the housing 12 has been moved axially with respect to the sleeve 14, causing the stationary ring 26 to push against and press the rotary seal 20, so that the rotary's forward edge 75 comes into slightly greater contact with the stationary element 26. At this point, the spacer clip members 46 are moved into position so that the end flange 42 on the housing 12 fits into the transverse groove 48 in each clip member 46, the forward ridge portion 41 of the clip member 46 fits into the space 44, and its other ridge 43 fits into a space 81 between the end ring 50 and the radial flange 42. With each spacer clip member 46 in position, the cap screws 54 are attached to hold them firmly in place. As the screws are tightened, the sleeve 14 is centered precisely within the gland housing 12.

The sealing device 10 as shown in Figs. 1 and 2, with spacer clip members 46 attached to hold the precise pre-load on the rotary, can be shipped and stored in such condition for ultimate use. When it is necessary to install a sealing device 10, it is first located on a pump shaft 60 and, as shown in Fig. 3, the housing flange 12 is

-14-

attached to the pump casing or stuffing box by the bolts 66. At this point, the set screws 58 (Fig. 3) in the holes 56 are advanced to secure the sleeve 14 firmly on the pump shaft 60. The pacer clip members 46 are then removed by withdrawing their cap screws 54.

5 Since the sleeve 14 is now secured to the pump shaft 60 and the housing 12 is secured to the pump casing 68, the bias or pre-load on the rotary seal 20 previously held by the clips 46 is maintained. The pump can now operate normally to develop the sealing surface 24 and seat it against the stationary seal 26 with no leakage between
10 the rotary and stationary elements 20 and 26.

In the sealing device 10 shown in Figs. 1-3, the sleeve 14 is preferably made of a low friction but durable plastic such as polytetrafluoroethylene (PTFE). Such an arrangement is particularly desirable where various deleterious fluids such as acid solutions
15 are being pumped.

For other applications, modified forms of the sealing device 10 may be provided within the scope of the invention. In the embodiment of Fig. 11, a sealing device 82 has a sleeve 84 made of metal, such as stainless steel, so that the additional metal end
20 ring 50 (Figs. 2 and 3) used on the device 10 is not required. In this embodiment an annular gasket 86 is partially recessed into the face of the housing flange 88 in lieu of the gasket 17 for device 10. Also, the separate stationary ring seal 26 is
25 eliminated. In this embodiment, the rotary seal bears against an annular bearing surface 90 which is provided in a recessed area of the housing. The bearing surface 90 is provided with a smooth hardened surface for receiving the forward sealing face of the rotary.

In the embodiment of Fig. 12, a sealing device 92 has a metal
30 sleeve 94 similar to sleeve 84, but a stationary ring 96 for contacting the rotary seal 20 is retained within an annular seat 98 that is formed within the housing 100. An O-ring 102 is provided adjacent the stationary ring seal to prevent any leakage around its

-15-

outer surface. An annular gasket 104 is also used to seal the inner surface of the housing 100.

5 With both the embodiments of Fig. 11 and 12, the pre-load bias of the rotary seal 20 is accomplished, as previously described by means of the spaced clip members 46 which are attached to the sleeve 84 or 94 and hold the housing and/or its stationary ring against the rotary seal, thereby providing the bias distortion of the rotary seal bearing surface against the stationary seal ring 96 or the housing surface 90.

10 In the embodiments of the invention shown above, the sealing devices 10, 82 and 92 are attached to the outside of a pump housing or stuffing box, as described. In such an arrangement the pump shaft is accessible through an opening in the stuffing box so that the sealing device can be installed on the shaft and secured to the stuffing box while maintaining the "pre-load" relationship between the rotary and stationary sealing elements, as described.

15 Fig. 13 shows a reverse type arrangement, as required for example, in sealing the shaft of a centrifuge. A shaft 110 is only accessible from the end of the sleeve which is nearest the rotary, that is inside the centrifuge or pressurized working area. In this arrangement, a sealing device 112 is provided which has an annular gland-housing 114 with an outer enlarged flange portion 116 which provides a circular space 118 and 119 around the shaft. A series of spaced apart bolts 120 extend through the enlarged flange portion 20 116 to secure it to the centrifuge structure 120 around the shaft. The flange portion 116 is integral with a central annular portion 122 of the housing 114 having a circular opening 119. Extending through the opening 119 of the gland housing 112 is a sleeve 124 which in this embodiment is preferably made of stainless steel, as previously described. The sleeve 124 fits around the shaft 110 and 25 has a rotary seal 136 that is supported within an outer circumferential groove 128 as with sleeve 14 in Fig. 2. At its distal end the sleeve 124 has an enlarged integral flange portion 130 centered within the circular space 118 around the shaft and is 30

-16-

spaced from the central annular portion 132 of the gland housing 114. Attached to the inner cylindrical edge of the annular portion 132 is a stationary ring 134 having a frontal surface that contacts a yieldable surface 136 of the rotary. A sealing O-ring 138 is
5 provided in a groove in the annular housing portion 132 that surrounds the stationary seal ring 134.

Spaced from the rotary seal 136 at the proximal inner surface of the sleeve 98 is an O-ring 140 which is confined in a groove 142. Spaced between the latter groove 142 and the proximal end 143 of the
10 sleeve 124 are a series of radial, threaded holes 144 for the set screws 146. These set screws hold the sleeve on the shaft after the desired pre-load has been accomplished. Extending through the central annular portion 132 of the housing 114 are a series (e.g. 4) of circumferentially spaced apart threaded holes 148. Each of these
15 holes 148 is adapted to receive a temporary pre-load adjustment pin 150 having a threaded portion 152 at its head end and a longer smooth pin portion 154 extending from the threaded portion.

When installing the sealing device 112, the rotary is in its relaxed state and the sleeve flange 130 is in contact with the inner
20 surface 156 of the housing 114. To "pre-load" the rotary seal 136, that is to force its sealing surface 137 into its partial surface contact with the outer face of the stationary ring seal 134, the sleeve 130 must be moved axially relative to the housing 114. This is accomplished by threading the adjustment pins 150 into the holes
25 148 so that their smooth portion 154 engages holes 158 in the sleeve flange 130 and pushes against the sleeve flange. Once the sleeve 130 is moved by the pins 150 the required amount to "pre-load" the rotary seal 136, the set screws 146 are advanced within the sleeve against the shaft 110 to hold the sleeve firmly in place. Now, the
30 pins 150 can be removed from the holes 148 and replaced with threaded plugs 160. Here again, rotary seal 136 and the stationary ring 134 provide a seal that prevents any migration of liquid or slurry through the housing 114 of the sealing device 112.

-17-

5 The arrangement in Fig. 13 using the adjusting pins 150 enables the sealing device 112 to be installed in otherwise inaccessible locations, and the pins 150 also enable the device to be centered properly so that it will operate efficiently with no leakage and a minimum of wear.

10 From the foregoing, it is apparent that the present invention provides a solution to the shaft sealing problem particularly for pumps required to handle slurries and other chemical mixtures which heretofore required large, continuous flow rates of fresh water. The present invention maintains effective sealing during shaft rotation and high pumping pressures without the need for any water purging.

15 To those skilled in the art to which this invention relates, many changes in construction and widely differing embodiments and applications of the invention will make themselves know without departing from the spirit and scope of the invention. The disclosure and the description herein are purely illustrative and are not intended to be in any sense limiting.

I CLAIM:

1. A unseated dry face seal for preventing leakage around a rotary shaft extending through a structural casing comprising
 - an annular stationary rigid seal means secured to the structural casing for surrounding the rotary shaft, the stationary rigid seal means having a smooth sealing surface means for establishing a dry face seal,
 - a resilient rotary seal means adapted to be secured to the rotary shaft and having an annular projecting edge means for contacting the smooth sealing surface under a biased pressure to form a dry face seal in conjunction therewith, and
 - the resilient rotary seal means comprising a carbon filled thermally resistant polymer composition having a durometer hardness of from about 85 to about 91.
2. An unseated dry face seal of Claim 1 wherein the thermally resistant polymer composition comprises a fluorocarbon filled with finely divided carbon black.
3. A dry face seal of Claim 1 wherein the annular contacting edge means of the resilient rotary seal means is a flat surface having a surface area greater than the contact area of the annular contacting edge means before seating occurs.
4. A dry face rotary seal of Claim 1 wherein the resilient rotary seal means comprises a resilient annular ring having a inner base portion adapted to be securely held in an annular mounting groove, the ring having a pivot point on a distal surface at the outer edge of the base portion, the proximal surface of the annular ring comprising outer and inner concentric proximal surfaces which converge to form an annular proximal sealing edge, the inner proximal surface extending from the sealing edge to a torque point on the proximal distal surface of the base portion.
5. A dry face rotary seal of Claim 4 wherein the angle formed by a line from the pivot point and the torque point in a plane

-19-

through the central axis of the seal is within the range of about 16 to 22°.

6. A dry face rotary seal of Claim 5 wherein the angle is within the range of from 17 to 21°.

- 5 7. An unseated dry face rotary seal of Claim 5 wherein the outer concentric proximal surface forms an angle of from about 40 to 50° with a line parallel to the central axis of the seal in a plane through said central axis, and the inner concentric proximal surface forms an angle of from about 55 to 73° with a line parallel to the central axis of the seal in a plane through said central axis.

- 10 8. An unseated dry face rotary seal of Claim 4 having radially outward, a forwardly diverging distal surface extending from the pivot point, and the shortest distance T, in millimeters, between the torque point and the distal surface in a plane through the central axis of the seal is provided by the formula:

$$T = \alpha + \beta ID$$

wherein α is from 3.2 to 3.6,

20 β is from 0.040 to 0.052, and

ID is the inner diameter of the proximal sealing edge, in millimeters.

9. A dry face rotary seal of Claim 4 wherein the distance D, in millimeters, of a straight line normal to the central axis of the seal from the proximal sealing edge to the surface of an axially concentric cylindrical surface through the torque point is provided by the formula:

$$D = \gamma + \delta ID$$

wherein γ is from 1.7 to 2.5,

30 δ is from 0.014 to 0.021, and

ID is the inner diameter of the proximal sealing edge, in millimeters.

-20-

10. A rotary seal element comprising a resilient annular ring having a inner base portion adapted to be securely held in an annular mounting groove, the ring having a pivot point on a distal surface at the outer edge of the base portion, the proximal surface of the annular ring comprising outer and inner concentric proximal surfaces which converge to form an annular proximal sealing edge, the inner proximal surface extending from the sealing edge to a torque point on the proximal distal surface of the base portion, and the seal element comprising a resilient, heat resistant carbon filled polymer composition having a durometer hardness of from 85 to about 91.
11. A rotary seal element of Claim 10 wherein the angle formed by a line from the pivot point and the torque point in a plane through the central axis of the seal is within the range of about 16 to 22°.
12. A rotary seal element of Claim 11 wherein the angle is within the range of from 17 to 21°.
13. A rotary seal element of Claim 10 wherein the outer concentric proximal surface forms an angle of from about 40 to 50° with a line parallel to the central axis of the seal in a plane through said central axis, and the inner concentric proximal surface forms an angle of from about 55 to 73° with a line parallel to the central axis of the seal in a plane through said central axis.
14. A rotary seal element of Claim 10 having a radially outward, forwardly diverging distal surface extending from the pivot point, wherein the shortest distance T, in millimeters, between the torque point and the distal surface in a plane through the central axis of the seal is provided by the formula:

$$T = \alpha + \beta ID$$

wherein α is from 3.2 to 3.6,

β is from 0.040 to 0.052, and

-21-

ID is the inner diameter of the proximal sealing edge, in millimeters.

15. A rotary seal element of Claim 10 wherein the length of a straight line D, in millimeters, normal to the central axis of the seal from the proximal sealing edge to the surface of an axially concentric cylindrical surface through the torque point is provided by the formula:
- 5
- $$D = \gamma + \delta ID$$
- wherein γ is from 1.7 to 2.5,
- 10 δ is from 0.014 to 0.021, and
- ID is the inner diameter of the proximal sealing edge, in millimeters.
16. An rotary seal element of Claim 10 wherein the thermally resistant polymer composition comprises a fluorocarbon filled with finely divided carbon black.
- 15

1/7

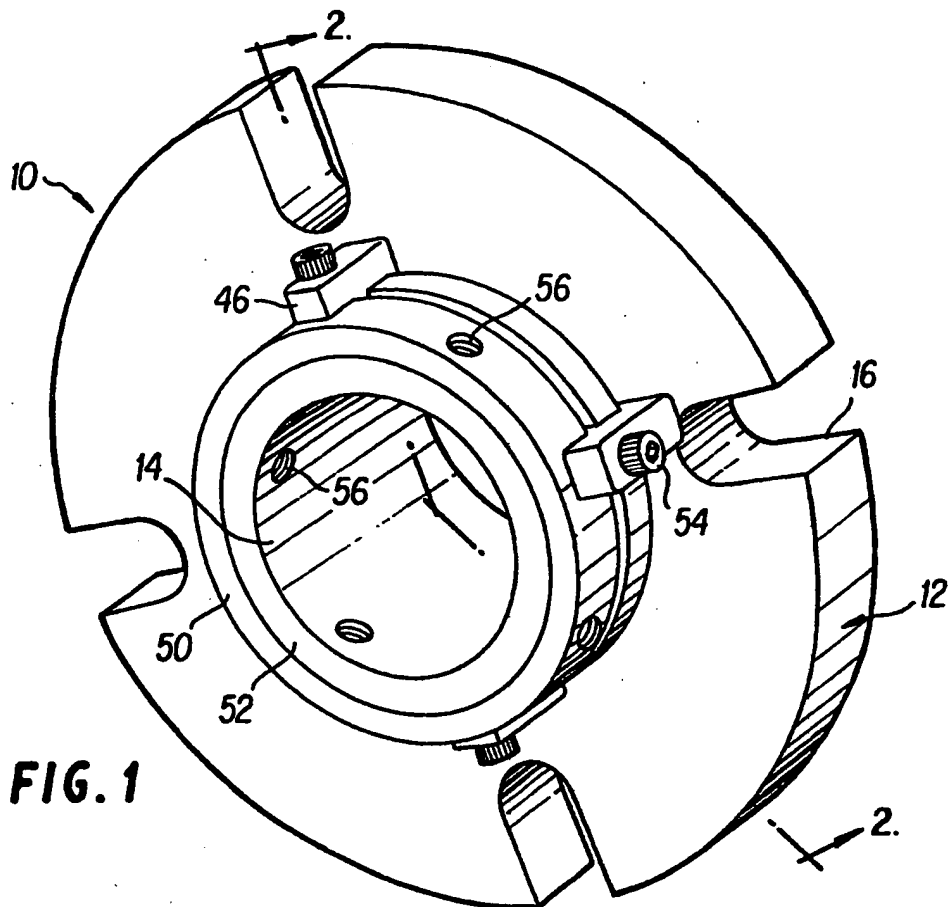


FIG. 1

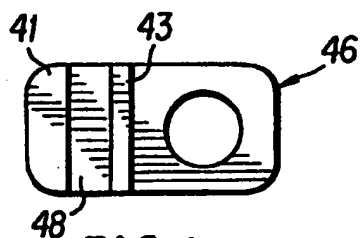


FIG. 4

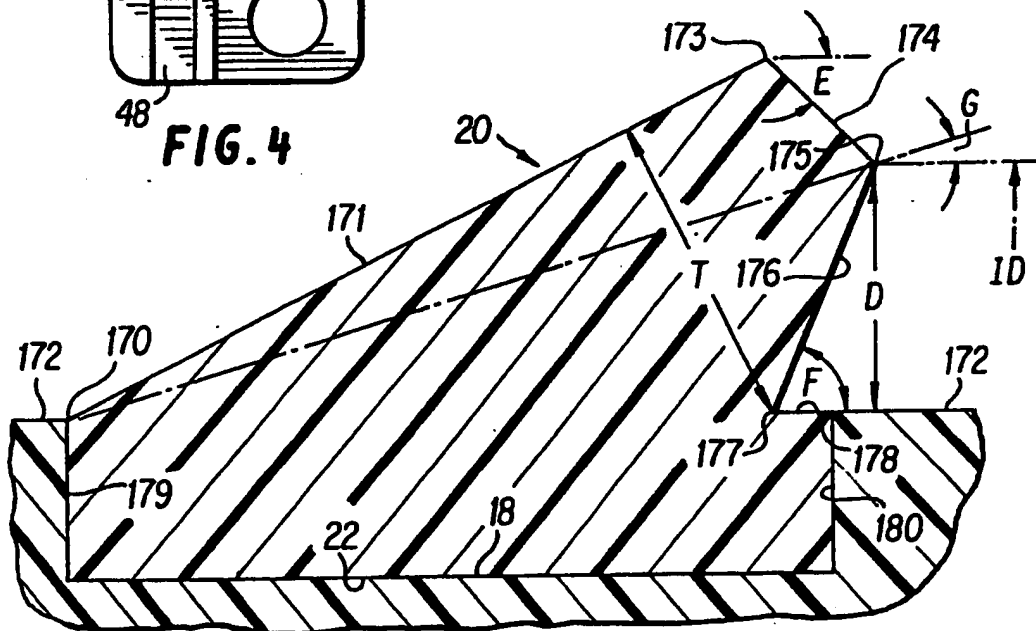
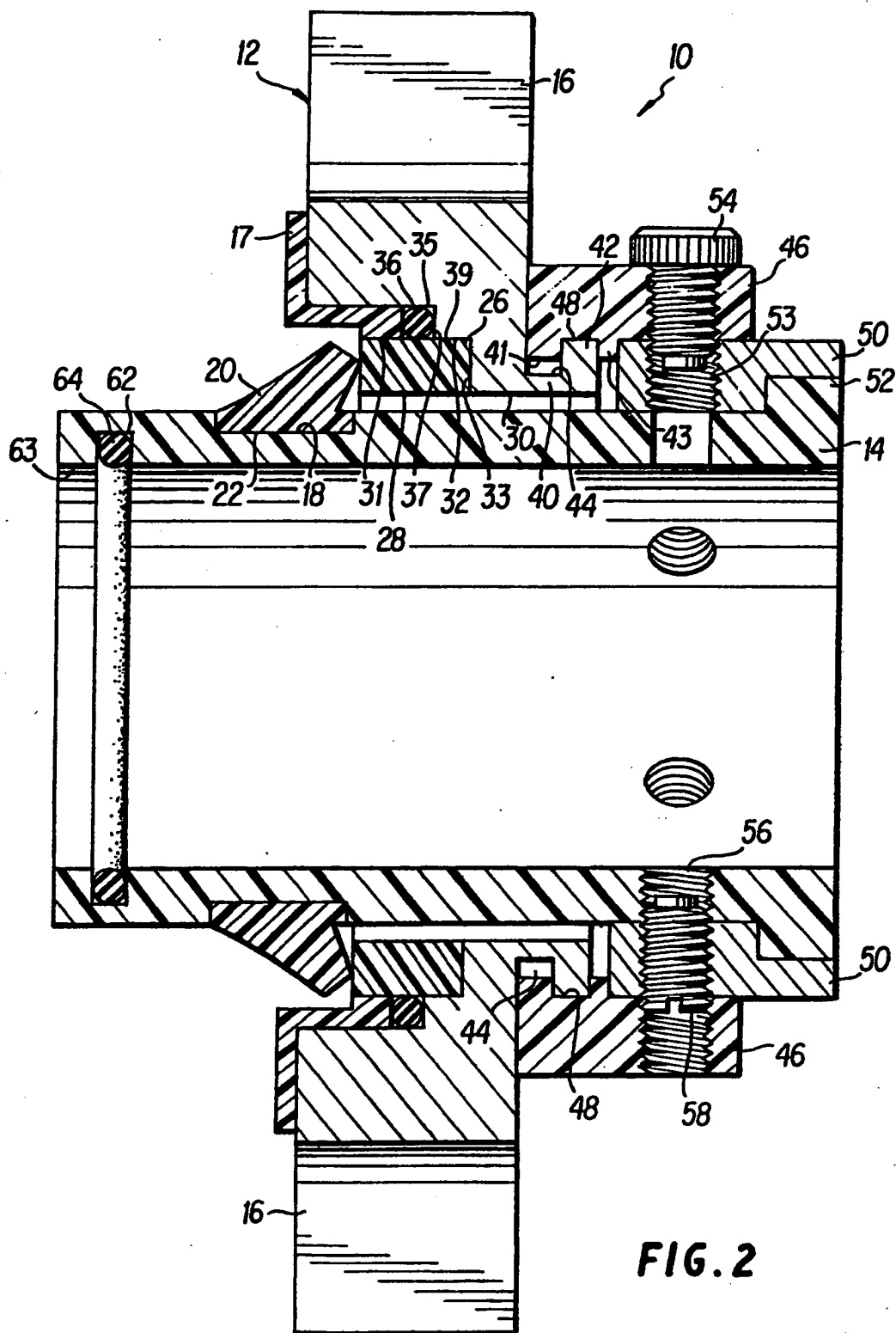
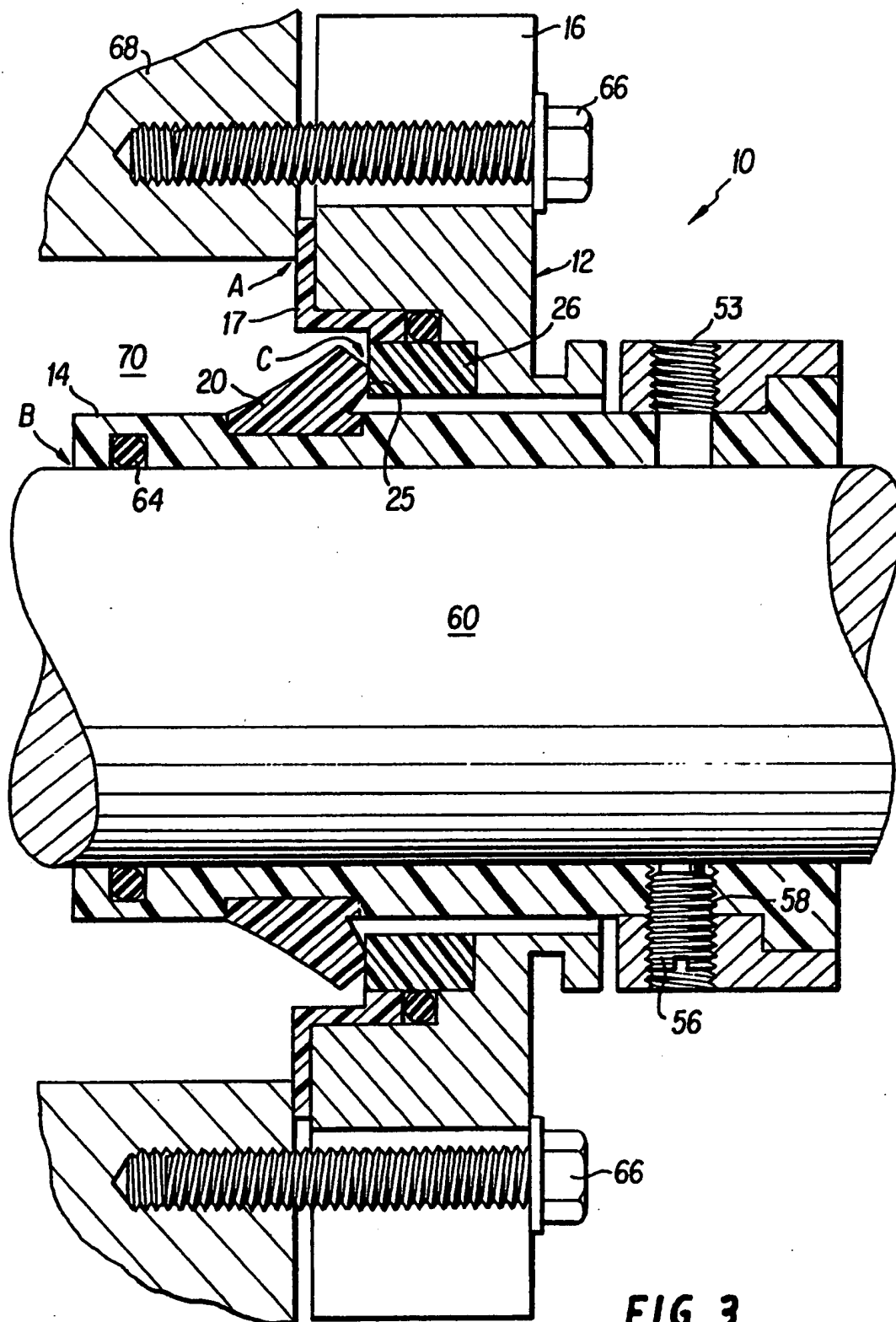


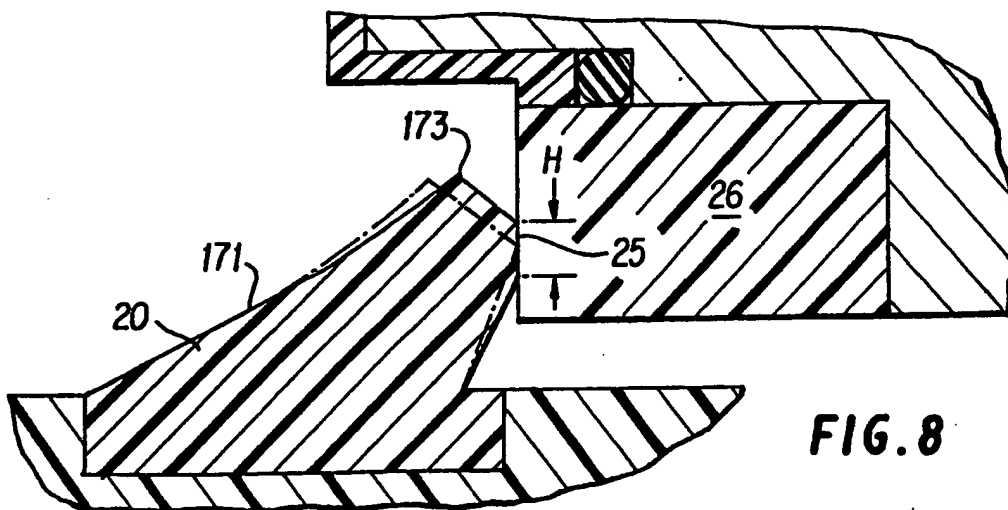
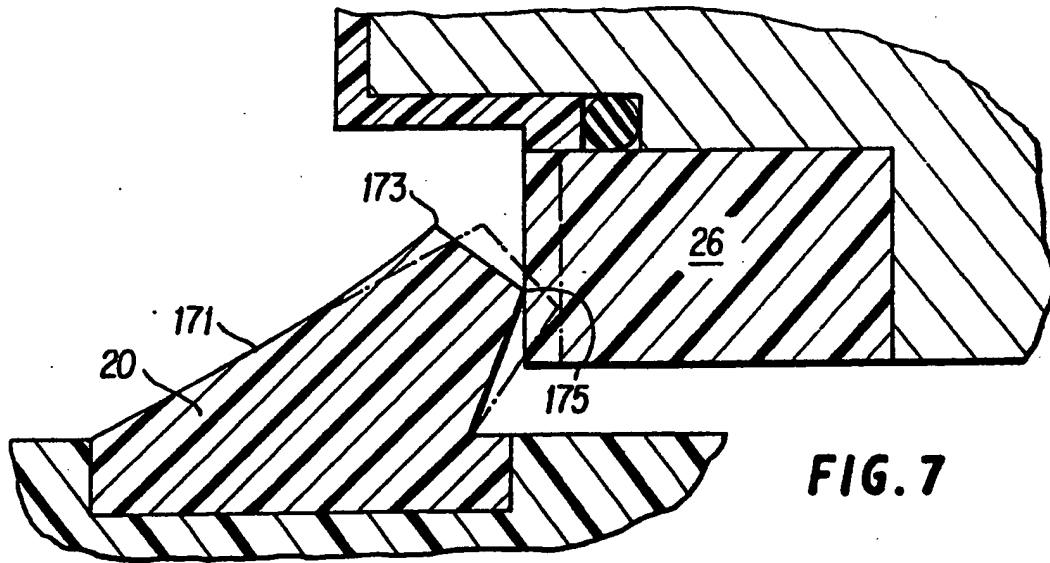
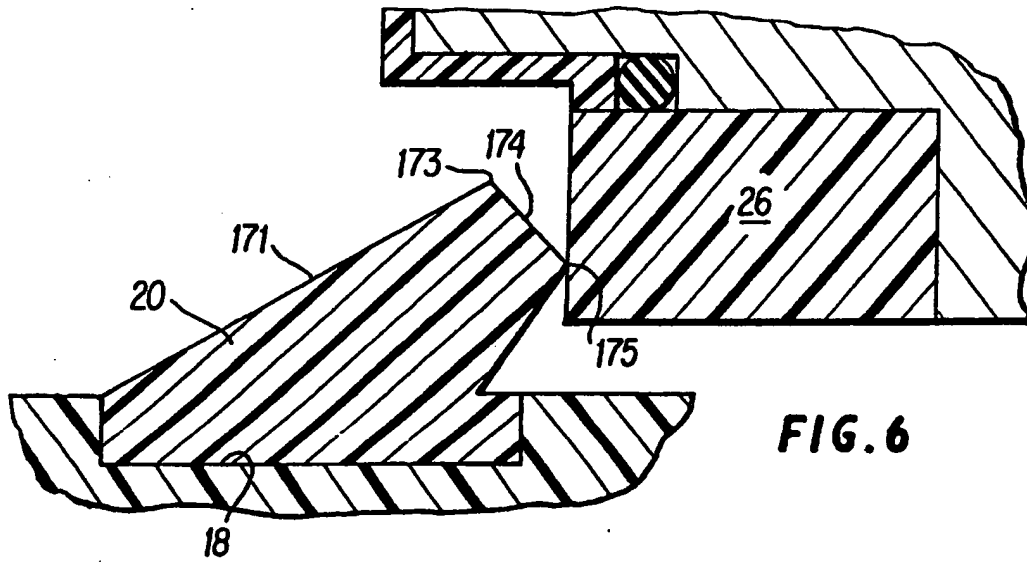
FIG. 5

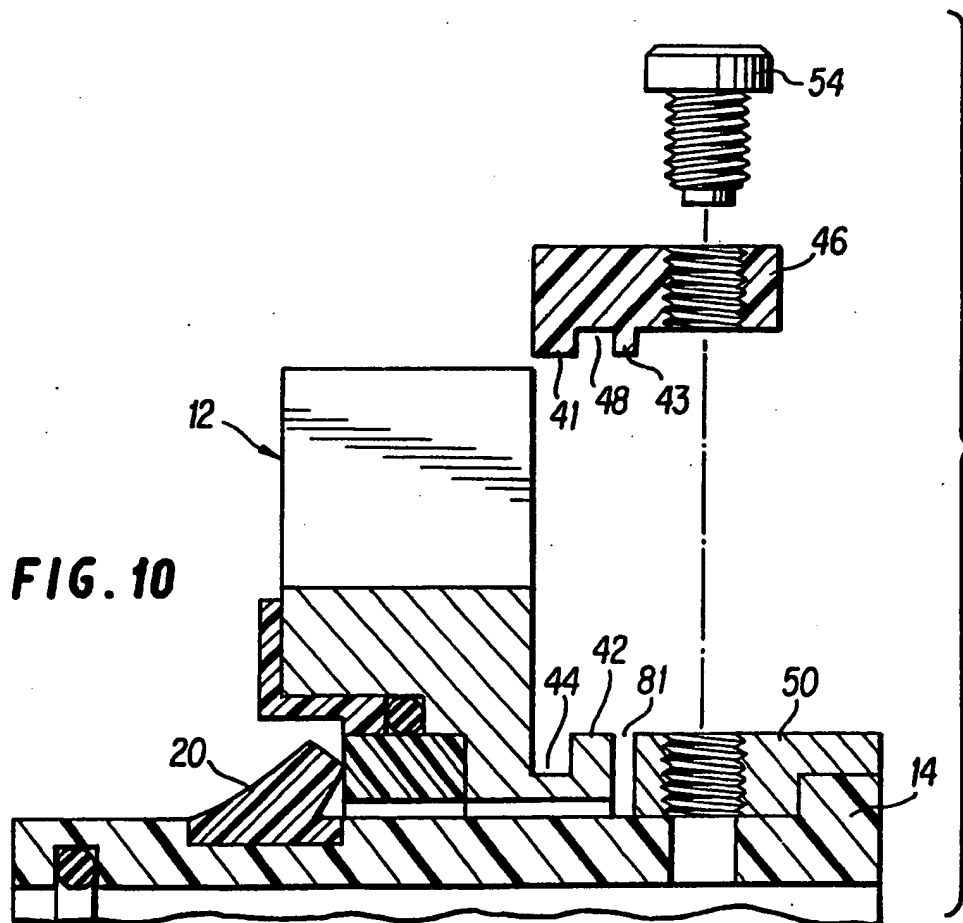
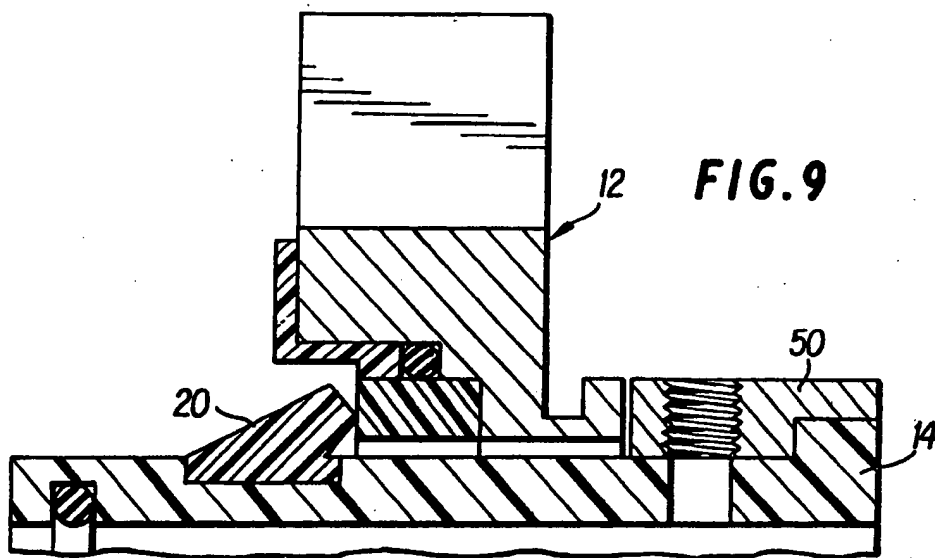
2/7



SUBSTITUTE SHEET







6/7

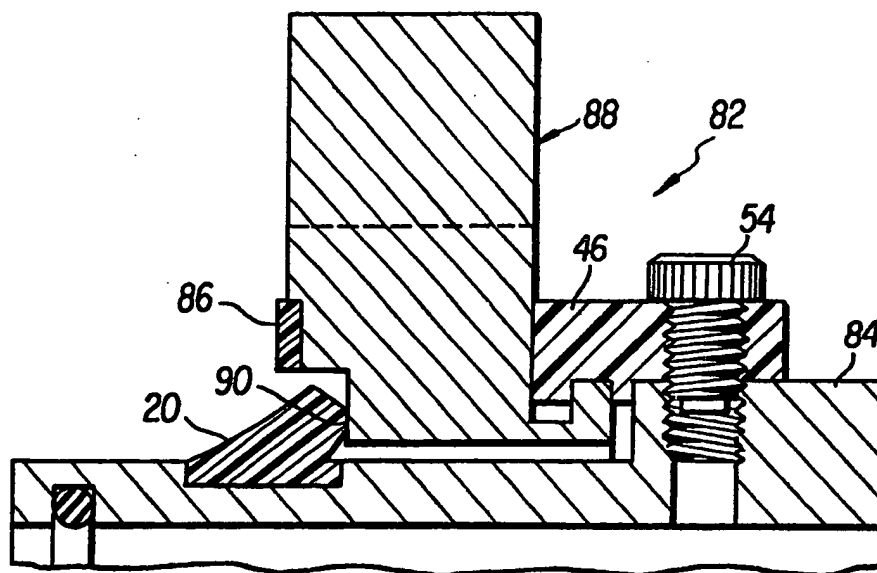


FIG. 11

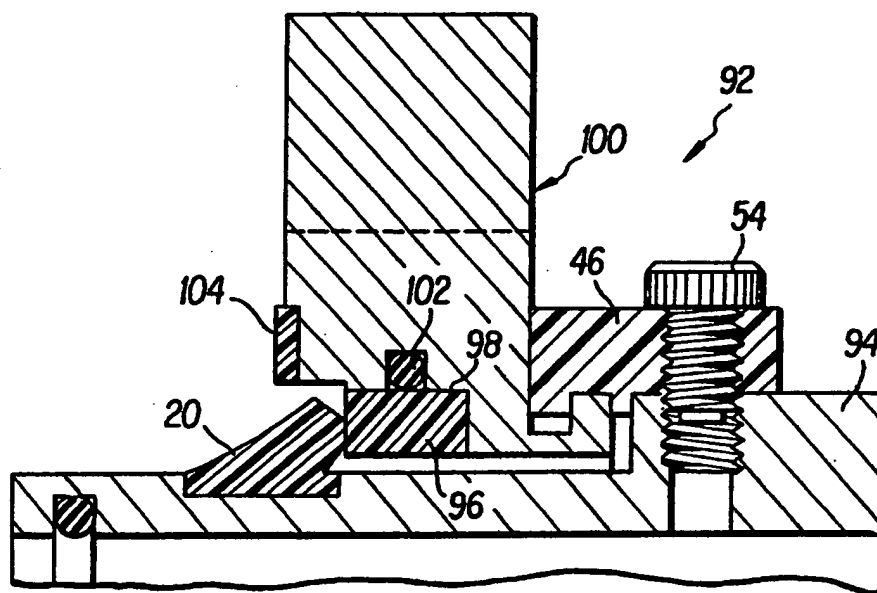


FIG. 12

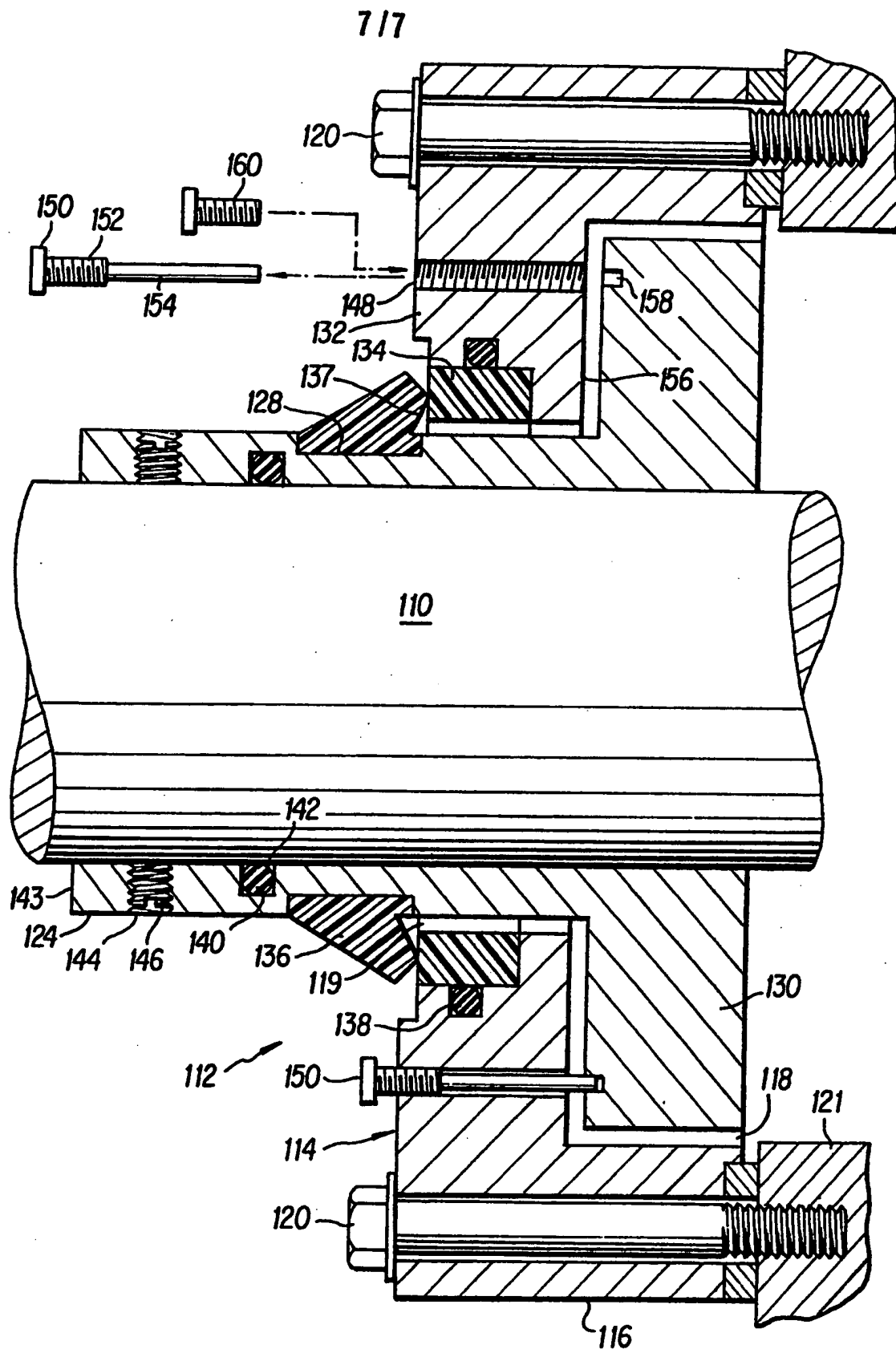


FIG. 13

SUBSTITUTE SHEET

INTERNATIONAL SEARCH REPORT

International Application No. PCT/US91/04987

| | | |
|--|--|---|
| I. CLASSIFICATION OF SUBJECT MATTER (If several classification symbols apply, indicate all) * | | |
| According to International Patent Classification (IPC) or to both National Classification and IPC | | |
| IPC(5): F 15/34 | | |
| U.S. CL: 2,7/95, 277/38 | | |
| II. FIELDS SEARCHED | | |
| Minimum Documentation Searched ⁷ | | |
| Classification System | Classification Symbols | |
| US. CL. | 277/9, 9.5, 11, 38-43, 64, 65, 81R, 95, 96, 96.2 | |
| Documentation Searched other than Minimum Documentation to the Extent that such Documents are Included in the Fields Searched * | | |
| III. DOCUMENTS CONSIDERED TO BE RELEVANT * | | |
| Category * | Citation of Document, ¹¹ with indication, where appropriate, of the relevant passages ¹² | Relevant to Claim No. ¹³ |
| A | US,A 4,973,063, Korenblit, 27 Nov. 1990 (see entire document) | 1-3 |
| A | US,A, 4,872,689, Drumm 10 October 1989 (see column 5, lines 39-59) | |
| A | US,A, 4,691,927, Sudol et al 8 September 1987 (see entire document) | |
| Y | US,A, 4,643,439, Lewis et al, 17 February 1987 (see entire document) | |
| A | US,A, 4,538,820 Duffee 3 September 1985 (see entire document) | |
| A | US, A, 4,413,830, Pietsch 8 November 1983 (see entire document) | |
| Y | US, A, 4,311,315, Kronenberg 19 January 1982 (see entire document) | 1-3 |
| <p>* Special categories of cited documents: ¹⁰</p> <p>"A" document defining the general state of the art which is not considered to be of particular relevance</p> <p>"E" earlier document but published on or after the international filing date</p> <p>"L" document which may throw doubts on priority claim(s) or which is cited to establish the publication date of another citation or other special reason (as specified)</p> <p>"O" document referring to an oral disclosure, use, exhibition or other means</p> <p>"P" document published prior to the international filing date but later than the priority date claimed</p> <p>"T" later document published after the international filing date or priority date and not in conflict with the application but cited to understand the principle or theory underlying the invention</p> <p>"X" document of particular relevance; the claimed invention cannot be considered novel or cannot be considered to involve an inventive step</p> <p>"Y" document of particular relevance; the claimed invention cannot be considered to involve an inventive step when the document is combined with one or more other such documents, such combination being obvious to a person skilled in the art.</p> <p>"A" document member of the same patent family</p> | | |
| IV. CERTIFICATION | | |
| Date of the Actual Completion of the International Search | | Date of Mailing of this International Search Report |
| 19 December 1991 | | 23 JAN 1992 |
| International Searching Authority | | Signature of Authorizing Officer |
| ISA/US | | Scott Cummings |

FURTHER INFORMATION CONTINUED FROM THE SECOND SHEET

| | | |
|---|--|-----------|
| Y | US,A, 4,295,654, Kawamura et al 20 October 1981 (see columns 1 and 3) | 1-3 |
| A | US,A, 4,136,887, Wentworth, Jr, 30 January 1979 see entire document | |
| A | US,A, 3,705,728, Miller 12 December 1972 (see entire document) | |
| A | US, A 3,112,113, taylor 26 November 1963 (see entire document | 4-9,11-15 |

V. ☐ OBSERVATIONS WHERE CERTAIN CLAIMS WERE FOUND UNSEARCHABLE¹

This international search report has not been established in respect of certain claims under Article 17(2) (a) for the following reasons:

1. ☐ Claim numbers _____, because they relate to subject matter¹² not required to be searched by this Authority, namely:
2. ☐ Claim numbers _____, because they relate to parts of the international application that do not comply with the prescribed requirements to such an extent that no meaningful international search can be carried out¹³, specifically:
3. ☐ Claim numbers _____, because they are dependent claims not drafted in accordance with the second and third sentences of PCT Rule 6.4(a).

VI. ☐ OBSERVATIONS WHERE UNITY OF INVENTION IS LACKING¹

This International Searching Authority found multiple inventions in this international application as follows:

1. ☐ As all required additional search fees were timely paid by the applicant, this international search report covers all searchable claims of the international application.
2. ☐ As only some of the required additional search fees were timely paid by the applicant, this international search report covers only those claims of the international application for which fees were paid, specifically claims:
3. ☐ No required additional search fees were timely paid by the applicant. Consequently, this international search report is restricted to the invention first mentioned in the claims; it is covered by claim numbers:
4. ☐ As all searchable claims could be searched without effort justifying an additional fee, the International Searching Authority did not invite payment of any additional fee.

Remark on Protest

- ☐ The additional search fees were accompanied by applicant's protest.
- ☐ No protest accompanied the payment of additional search fees.

| III. DOCUMENTS CONSIDERED TO BE RELEVANT (CONTINUED FROM THE SECOND SHEET) | | |
|--|---|----------------------|
| Category * | Citation of Document, with indication, where appropriate, of the relevant passages | Relevant to Claim No |
| A | US,A, 3,072,414, Porges 8 January 1963 (see entire document) | 1-3 |
| A | US, A, 3,015,505, Smith 2 January 1962 (see entire document) | |
| Y | US,A, 2,863,680, Taltavall, Jr. 9 December 1958 (see column 3, lines 32-59) | |
| A | US,A, 2,077,954, Ramclow 20 April 1937 (see entire document) | |
| A | CA, A, 671,238, Yist 24 September 1963 (see entire document) | |
| A | JP, A, 118,377, 14 July 1983 (see trans. abstract) | |
| A | JP,A, 118,375, 14 July 1983 (see trans. abstract) | |
| A | CR Seals Handbook, catalog #457010 January 1986, (see pages 3, 4, 10 and 25) | 1-3 |
| Y | Mechanical Face Seal Handbook published by Chilton Book Co. August 1975 (see pages 76-77) | |